Proceedings of the 1998 U.S. DOE Hydrogen Program Review

NREL/CP-570-25315

Internal Combustion Engine Report:
Spark Ignited ICE GenSet Optimization
And
Novel Concept Development

1998 Hydrogen Technical Review

by

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Abstract

In this manuscript we report on two projects each of which the goal is to produce cost effective hydrogen utilization technologies. These projects are: 1) the development of an electrical generation system using a conventional four-stroke spark-ignited internal combustion engine generator combination (SI-GenSet) optimized for maximum efficiency and minimum emissions, and 2) the development of a novel internal combustion engine concept. The SI-GenSet will be optimized to run on either hydrogen or hydrogen-blends. The novel concept seeks to develop an engine that optimizes the Otto cycle in a free piston configuration while minimizing all emissions. To this end we are developing a rapid combustion homogeneous charge compression ignition (HCCI) engine using a linear alternator for both power take-off and engine control.

This year our SI-GenSet program shifted from a research and development activity to a technology illustration and deployment activity. The goal of our redirected effort is to produce an optimized internal combustion engine generator set (SI-GenSet) using our current knowledge of hydrogen combustion in an engine. Because of this shift in emphasis, the fluid dynamic reacting flow modeling and the chemical kinetic modeling activities, previously associated with this program, were not funded. A secondary goal is to develop alliances with automobile OEM's (Original Equipment Manufacturing) to promote the utilization of this technology.

The target design for the optimized, spark-ignited internal combustion engine is an indicated thermal efficiency (ITE) of 47% with emissions below 5 PPM of NO_x. It is estimated that with this target the SI-ICE will yield an overall brake efficiency of 40% and emissions of NO_x below the Super Ultra-Low Emission Vehicle (SULEV) or the Equivalent Zero Emission Vehicle (EZEV) proposed legislation by the California Air Resources Board (CARB). Indeed, combining our engine out values with standard NO_x reduction technologies it is shown that this technology will produce emissions in the 0.13 PPM range (below ambient conditions in some cities).

In helping to achieve the goal of technology transfer, we have established alliances with two domestic OEMs. Both of these organizations have expressed interest in developing hydrogen-fueled ICE's based on our work.

Our rapid combustion homogeneous charge compression ignition (HCCI) free piston project has demonstrated indicated thermal efficiencies of 56% with a estimated overall efficiency of 50% while maintaining NO_x in the few PPM level. One side benefit from this HCCI technology is fuel flexibility. Fuel flexibility can have a significant beneficial impact facilitating the transition from hydrocarbon fuels to hydrogen.

Introduction

In this manuscript we report on two projects: 1) the development of a conventional four-stroke internal combustion engine generator set (SI-GenSet) optimized for maximum efficiency and minimum emissions, and 2) the development of a novel internal combustion engine concept. The SI-GenSet can be configured to run on either hydrogen or hydrogen-blends. The novel concept seeks to develop an engine that optimizes the Otto cycle in a free piston configuration while minimizing all emissions. To this end we are developing a rapid combustion homogeneous charge compression ignition (HCCI) engine using a linear alternator for both power take off and engine control.

This year our SI-GenSet program shifted from a research and development activity to a technology illustration and deployment activity. The fluid dynamic reacting flow modeling and the chemical kinetic modeling activities previously associated with this program were not funded this year. We were directed to take a snapshot in time and produce an optimized spark ignited internal combustion engine generator set (SI-GenSet) utilizing our knowledge to date. In addition, we were to develop alliances with automobile original equipment manufactures (OEM's) to promote the utilization of this technology.

Our rapid combustion homogeneous charge compression ignition (HCCI) program is supported by internal Laboratory Directed Research and Development (LDRD) funds and is leveraged with funds from the alternative fuels program from the Office of Advanced Automobile Technologies (OAAT). As such, we will discuss progress in this project as it relates to both hydrogen and alternative fuels.

This report describes the motivation, approach, progress to date, and the parameters that will be employed in the design of our SI-GenSet. This report will also describe the progress made with our rapid combustion HCCI project and future plans.

Motivation

The objective of this program is to develop cost effective highly efficient and ultra-low emission hydrogen fueled end-use technologies for the immediate utilization. Targeted applications include stationary electrical power generation, stationary shaft power generation, hybrid vehicles, and nearly any other application now being accomplished with internal combustion engines.

It is interesting to note that hydrogen use can't be motivated by energy security and pollution concerns. For example, shifting to natural gas as an energy source solves our energy security problem (at least for the near term) and solves all of our current criterion gas emission problems. (Note: Honda has a commercially available natural gas vehicle that satisfies the proposed SULEV and EZEV emission levels in California.) The use of hydrogen as an energy carrier can actually increase the emission of CO_2 (a global climate change gas) over a conventional solution if attention to CO_2 is not given over the entire fuel cycle (cradle to grave). For example, a diesel fueled hybrid vehicle will emit less CO_2 than a hydrogen fueled hybrid vehicle if the hydrogen is made from electrolysis from electrical power made from fossil fuels (coal and/or natural gas fueled turbines) and the resulting CO_2 is not sequestered in a suitable fashion. (S. Thomas, Directed Technologies Inc., private communication May 1998).

Two motivators for the use of hydrogen as an energy carrier today are: 1) to provide a transition strategy from hydrocarbon fuels to a carbonless society and 2) to enable renewable energy resources. The first motivation requires a little discussion while the second one is self-evident. The most common and cost effective way to produce hydrogen today is the reformation of hydrocarbon fuels, specifically natural gas. Robert Williams [1] discusses the cost and viability of natural gas reformation with CO_2 sequestration as a cost-effective way to reduce our annual CO_2 emission levels. He argues that if a hydrogen economy was in place then the additional cost of natural gas reformation and subsequent CO_2 sequestration is minimal [1]. Decabonization of fossil fuels with subsequent CO_2 sequestration to reduce or eliminate our CO_2

atmospheric emissions provides a transition strategy to a renewable, sustainable, carbonless society. However, this requires hydrogen as an energy carrier.

If hydrogen is to make it into the market place the end-use technologies must satisfy a set of suitable boundary conditions while providing the market place a sound, cost-effective technology for end-use. The boundary conditions applied to this problem are:

1. Cost effective capital equipment.
2. The technology must be energy efficient to offset the cost of hydrogen production.
3. For transportation applications the power density must be high to be packaged in a vehicle (transportation is the most stringent application).
4. For transportation applications the specific power must be sufficiently high to keep the mass of the vehicle to a minimum (transportation is the most stringent application)
5. The technology must be environmentally benign (near zero (or below ambient) or zero emissions)
6. There must be compatibility with existing overall infrastructure including parts distribution, service and maintenance personnel. Infrastructure compatibility makes the introduction of a new technology easier.
7. The technologies must be implemented in a way to enable market penetration in the easiest way possible.
8. We need to implement technologies and transition strategies to grow the hydrogen infrastructure that will enable emerging technologies market penetration in the easiest way possible.

Figure 1 shows a comparison of the characteristics of different end-use technologies. The numbers presented are for the overall system needed to employ a particular technology. The fuel cell systems includes the stack, fluid handling equipment, necessary cooling hardware, etc. These numbers exclude any fuel processing equipment since that will be common to all hydrogen end-use technologies and is system dependent (i.e., reformation on or off site). We do consider the transportation market not because it represents the best opportunity for hydrogen use but because it represents the most technically challenging solution.

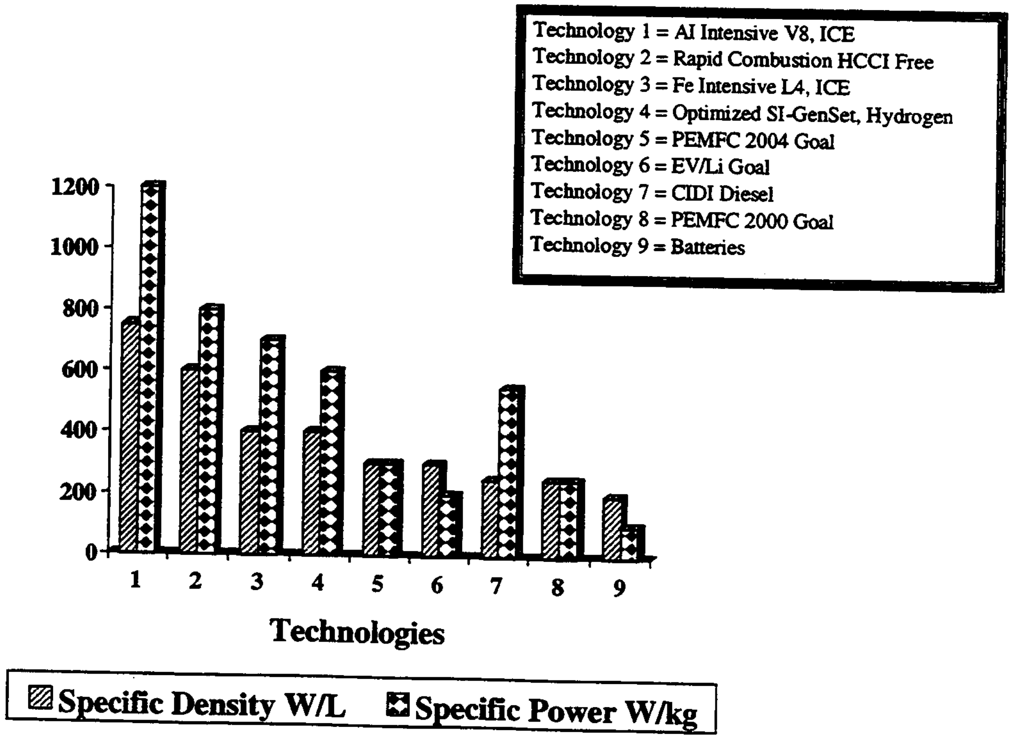


Figure 1. Power densities and specific power for selected energy conversion devices.

A conclusion from the data in Figure 1 is that an optimized ICE exceeds the power density and specific power DOE PEM system goals for the year 2004. Figure 1 compares only the power density and specific power, the system efficiencies need to be compared as well. Simply comparing system efficiencies between technologies without considering an appropriate driving cycle can be very misleading; a thorough analysis is beyond the scope of this work. However, a first order analysis was performed to determine appropriate efficiency targets for the SI-GenSet. Shown in Figure 2 are two vehicle schematics showing the power routing and the relevant efficiencies of each major component. A driving cycle was assumed so that 50% of the drive power goes through the base load system, 30% goes through the peaking system, and 20% is attributed to the regenerative breaking. The results of this first order analysis is given in Fig. 3. Fig. 3 shows that with an efficiency of ~40%, a contemporary optimized SI-GenSet powered hybrid vehicle is within 80% of the hybrid vehicle powered by a PEM fuel cell operating at the DOE Year 2004 goal. It should be noted that a much more sophisticated analysis of the same vehicle types analyzed here was performed by S. Thomas (private communication, May 1998). Thomas predicts 52.9 m/g for a vehicle with a gross vehicle weight (GVW) of 1247 kg for a hydrogen fueled SI-GenSet operated in the same parallel configuration as above and 66 for a vehicle with a GVW of 1291 kg for the fuel cell hybrid vehicle. A comparison using S. Thomas' results yields 52.9/66 = 80% which is the same as in our analysis. Thus the SI-GenSet system addressed in this work provides an attractive technology that satisfies all of the boundary conditions necessary to use hydrogen in a cost effective way today. Implementing optimized conventional technologies as is demonstrated here will pave the way for emerging technologies to enter the market as they mature, such as the emerging PEM fuel cell systems.

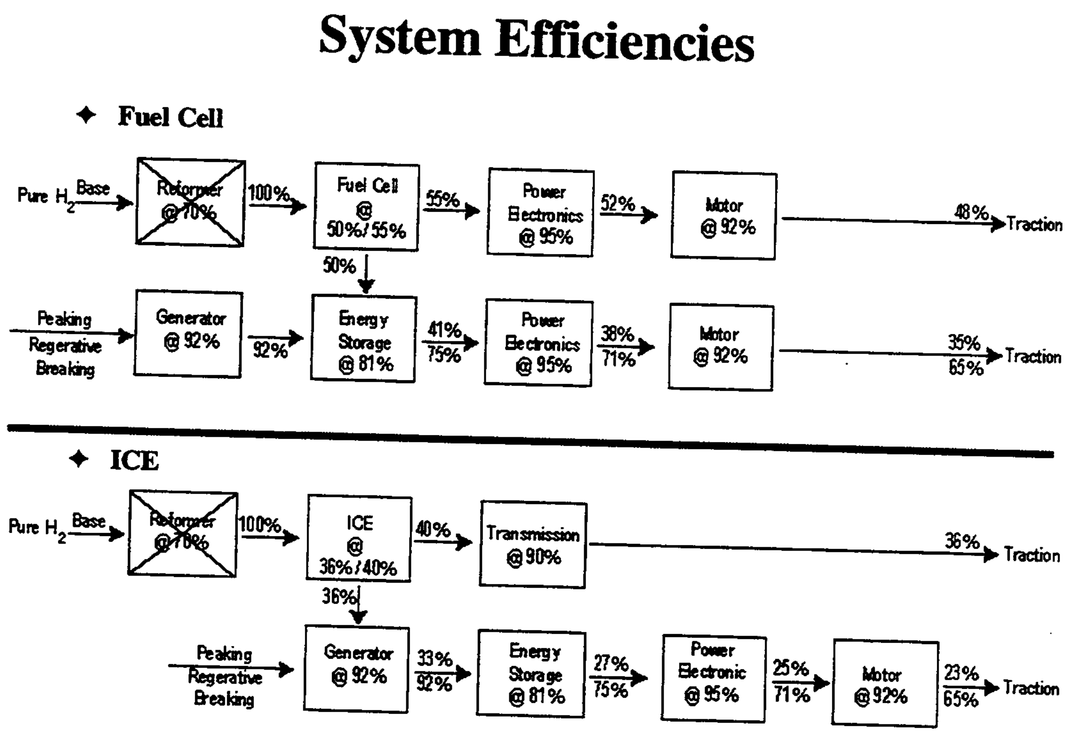


Figure 2. Vehicle system efficiency map.

Vehicle System Efficiencies - Score Card

- **DOE Fuel Cell System Goals**
  - 50% @ 100% load, 57% @ 25% load
    - Straight line gives
      - 50 % @ 100% load, 55% @ 50% load
- **ICE Goal**
  - 36% @ 100% load, 40% @ 50% load

Vehicle System Performance Estimate

	Fuel Cell	ICE	Why wait? (η_{ICE}/η_{FC})
Pure H_2	49 %	39 %	80 %

Figure 3. Comparison between a fuel cell powered hybrid vehicle and an optimized ICE powered hybrid vehicle

Development Progress to Date

In previous work we established the ability to successfully control the combustion process in an SI ICE yielding extremely low emissions (<5 PPM). Shown in Figure 4 are data taken from our research engine that shows a wide range of operating conditions all with engine-out NO_x levels sufficiently low to meet the California proposed EZEV regulation (< 0.02 gm/mile). Controlling the emissions in this way results in a system that does not rely on any post clean up technologies (i.e. a catalytic converter) and can not degrade with time.

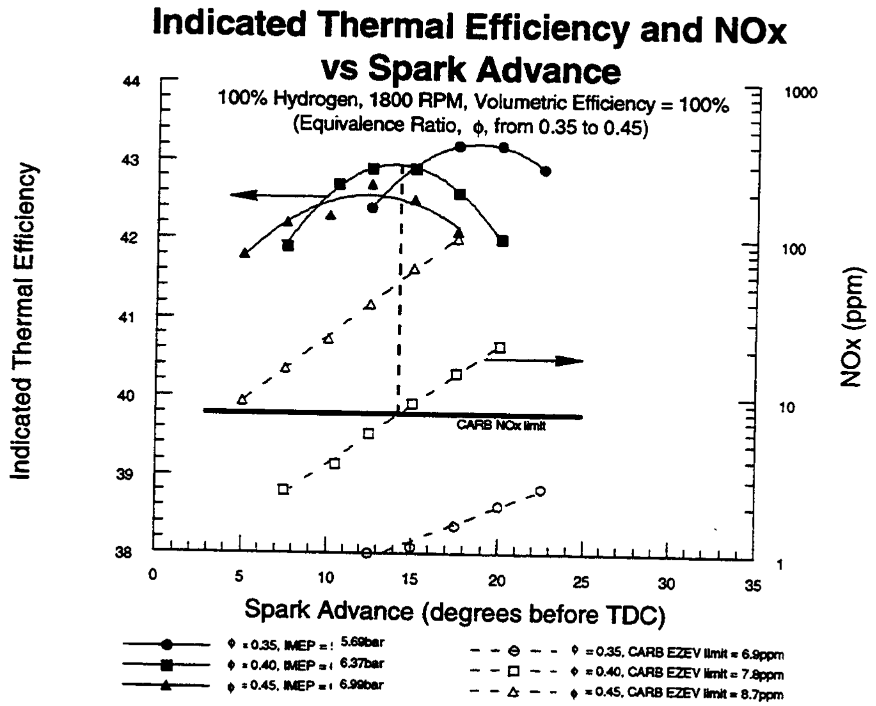


Fig 4. NO_x versus engine performance from our research engine

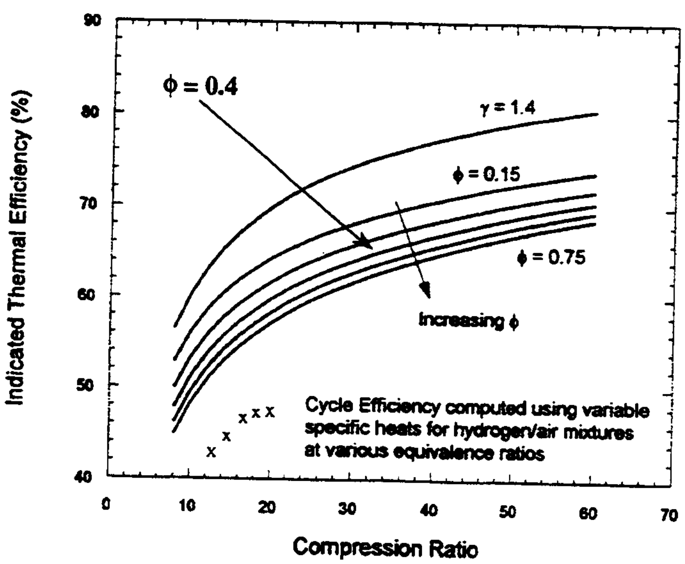


Figure 5. Measured indicated thermal efficiency (x) and theoretical indicated thermal efficiency (lines) as a funtion of compression ratio for several equivalence ratios.

Figure 5 shows the indicated thermal efficiency as a function of compression ratio compared to the theoretical maximum. The lines show the theoretical calculations and the points represent data from our current research engine. Like other researchers (Caris and Nelson (4)) we expect that with increased compression ratio the performance of our system will start to fall off and deviate from the shape of the theoretical values. While more work needs to be done to accurately predict the efficiency of our system the peak in efficiency most likely occurs at compression ratios of 18:1 to 20:1. We are in the process of obtaining data at 22:1 to further characterize the efficiency curve. For the current configuration, as shown in Figure 5, we have achieved an indicated efficiency of 47%, a value which is 80% of the theoretical maximum. Assuming parasitic losses of 15% (conventional wisdom for Ice's is 10% at full load (Heywood) yields a break efficiency of 40% which is our goal. Our previous work has shown that this particular combustion chamber configuration is not optimized for efficiency, so further improvements can be made by reducing the swirl and fine scale turbulence, and reducing the friction losses. Recall that because we operate lean premixed the emissions from this engine are controlled to be sufficiently low (or lower) to meet the current regulations (10% of ULEV or the proposed EZEV regulations in this case). Indeed, applying the same emission control physics but using exhaust gas recirculation instead of operating lean premixed one can obtain the same engine out NO_x values and can further reduce NO_x by adding a NO_x reduction three way catalysis to the exhaust system. Other researchers have demonstrated this approach and obtain similar engine out values to the work reported on here, but they further reduce their tailpipe emissions to about of 0.13 PPM in some cases. Recall that NO_x values in some of our cities on bad days exceeds this value (Riverside California on a bad day will reach values on the order of 0.3 PPM). Even though with this approach the NO_x levels are extremely attractive recent results suggest that N_2O is formed by the catalyst resulting in elevated levels out of the tailpipe. N_2O is a known global climate change gas that is about 355 times worst than is CO_2.

Operating a lean homogenous charge is our operating condition of choice. Operating lean provides a thermodynamically favorable condition resulting in increased efficiency and provides control over the engine out NO_x values. NO_x emissions can be controlled to sufficiently low values to satisfy the most stringent emission proposed regulation without any emission post clean up technologies thus eliminating a source of N_2O.

SI-GenSet Summary

This project is primed to move on to the next phase of development. We have successfully demonstrated sufficient control over the combustion process to guarantee engine-out NO_x concentrations sufficiently low to satisfy the most stringent of proposed regulations. Our engine out NO_x values would qualify a hybrid vehicle obtaining an equivalent 60 miles/gal to meet the proposed Equivalent Zero Emission Vehicle (EZEV). This can be compared to the Toyota Prius, which is reported to achieve 61 miles / gallon of gasoline. We have also demonstrated a sufficiently high indicated thermal efficiency that our estimated brake efficiency is at 40%. This value puts an ICE powered hybrid within 80% of the DOE PNGV goal for a PEM fuel cell system. We are now in the process of purchasing a multi-cylinder engine generator set that will yield ~50kW of peak output with an optimized base output of 25 kW. Based on our current knowledge base we expect to configure this engine to operate with the following characteristics:

1. 25 kW base load with a 50kW peaking capability. This allows us to optimize the engine for base load and take a minor degradation in efficiency under peak requirements. Note that this engine will operate at peak load only for the "infinite hill climb." All other peaking requirements will be met by the on-board energy storage device, of which batteries are the most likely choice.
2. Heat transfer losses will be minimized by configuring the engine with as large a volume to surface area ratio as is reasonably possible, and as such we will use a 3.2 liter, 4 cylinder engine block as our starting point.
3. The piston and head will be designed to minimize swirl and fine scale turbulence to keep the heat transfer losses to a minimum.
4. The valve timing will be determined in the final configuration to maximize the trade off between increasing the efficiency by turboboost or maximizing the energy extracted by the expansion stroke. This is necessary because of the trade off that occurs between extracting as much energy as possible during the expansion stroke versus leaving enough energy in the exhaust system for use by a turbo charger.

5. This engine will be set up to operate at a compression ratio of about 20:1. The exact value will depend on the outcome of our current measurements
6. At these very high compression ratios, and high diluent loading the burn duration becomes a limiting factor. To maximize the efficiency it is crucial to minimize the burn duration (to better approximate the theoretical maximum efficiency from the Otto Cycle). As such this engine will be equipped with a dual spark plug configuration.
7. The engine will be designed to operate at the nominal power output of ~25kW unthrottled, and at a RPM of about 1800 to 2400 depending on the SI-GenSet match. The high power output (only for the infinite hill climb) condition will be obtained by increasing the engine RPM and the pressure boost from the turbocharger as needed to match the load requirements. This will result in a slight loss in efficiency but it will result in no degradation in the emission performance. Since the duty cycle under these conditions is very low the overall performance of the vehicle will not suffer.
8. The key to guaranteeing the emission performance is to operate the combustion in a lean premixed (homogenous charge) configuration. At equivalence ratios of 0.35 to 0.40 we are well below the knee in the NO_x curve and our measured results are well below the values needed to satisfy the proposed EZEV regulations by CARB. The lean operating limit for these engines under similar conditions is about an equivalence ratio of 0.25. This provides a significant range in equivalence ratio to provide control over the combustion temperature due to weak changes as a result of changing RPM and/or pressure boost.
9. Fuel injection will be done in the port by conventional high pressure injection systems as perfected by the work of Scott, and Heffel at the University of California at Riverside College of Engineering Center for Environmental Research and Technology (CE-CERT) (private communication, P. Scott 1998). The injection will be timed for open valve conditions to provide a fluid dynamic mixing environment to ensure a premixed charge entering into the combustion chamber. This timing and the dilute operation will ensure that preignition does not occur.

Rapid Combustion HCCI Electrical Generator

The advanced generator concept being developed by Sandia utilizes a free piston engine configuration where the piston's oscillating motion directly generates electrical power. A linear alternator component is integrated into the device such that the varying magnetic flux of permanent magnets attached to the piston is harnessed. A schematic of the design is presented in Figure 6.

Within the free piston engine, combustion occurs alternately at each end of the cylinder, thus driving the piston motion. The operating compression ratio is controlled by the linear alternator which precisely manages the piston's kinetic energy through its stroke. A homogeneous charge compression ignition (HCCI) combustion process is employed where the cylinder's premixed fuel-air charge is compressed to the point of autoignition. Compression is achieved quickly, and rapid combustion occurs at nearly top dead center. A two-stroke cycle scavenging process is used to recharge the engine's cylinder.

Several advantages ensue from this unconventional engine design. First, the HCCI combustion process can be extremely fast; ensuring that most of the fuel energy is converted to heat at maximum compression. The Otto cycle's constant-volume combustion condition is more closely approached in this configuration, and thus the efficiency can be maximized. As a result, the engine's achievable thermal efficiency is not restricted by finite combustion processes such as flame propagation or mixing/diffusion times, as it is in conventional IC engines.

A second important attribute of the engine is that the free piston's compression ratio is variable, and potentially greater than in crankshaft-driven configurations. This characteristic allows very lean fuel-air mixtures (equivalence ratios near 0.3) to be successfully ignited at high compression ratio using the HCCI process and it enables operation on a variety of different fuels without significant hardware modifications. Lean operation dramatically reduces the formation of NO_x, and can improve the engine's thermal efficiency. High compression ratios (greater than 20:1) increase the efficiency of the engine cycle.

Finally, the unique dynamics of the free piston aid in minimizing the time that the cylinder gases spend at elevated combustion temperatures. Thus, the free piston configuration enables the heat losses from the cylinder charge to be reduced, and NO_x emissions to be controlled.

Integration of the linear alternator into the free piston geometry provides further benefits to the generator design. In this arrangement mechanical losses in the system are dramatically reduced since there is essentially one moving part, and this allows engine operation at a more or less constant piston speed. These points aid in the generator design, and further improve the fuel-to-electricity generation efficiency of the device.

Benefits of the Rapid Combustion Electrical Generator

The rapid combustion electrical generator represents a revolution in IC engine-generator technology as the efficiency of the device is significantly improved relative to current technologies, while the emissions can be satisfactorily controlled. It is expected that the fuel energy conversion efficiencies will represent an improvement of 20-50% over conventional IC engine-generators (including the class of micro-turbines) with the manufacturing costs comparable to today's piston engines. Additionally, the emissions controls, using only an oxidation catalyst when utilizing hydrocarbon or hydrocarbon / hydrogen blends should enable a small generator of this type to have emissions comparable to large stationary generating plants. Finally, the advanced generator would be able to operate on a multitude of fuels without significant reconfiguration. The capabilities of fuel diversity could allow for greater customer choice, an easy integration of renewables into the electrical system, and the flexibility of fuel substitution based on market considerations.

Technical Feasibility

As with any proposed revolution in technology, the feasibility of achieving the expected advances with the generator concept must be satisfactorily established. Through various background studies, as well as recent experimental and computational investigations, this task has been conducted.

Recent advances in the areas of IC engine combustion and gas transfer, electrical power generation, and modern control theory have addressed critical components necessary for the generator's development. Each of these individual components are established. The integration of the HCCI combustion system, the scavenging cycle, linear alternator, and control dynamics is the purpose of the ongoing project. The following discussion details the work to date, and fully establishes the motivation for future development.

<u>Combustion System</u> - Internal combustion engine systems operating on the Otto cycle ideally represent very high conversion efficiency machines. Thermodynamically, there does not appear to be any fundamental limit to the potential of these devices. Edison (3) analytically investigated the efficiency of the ideal Otto cycle at compression ratios of up to 300:1, including the effects of chemical dissociation, working fluid thermodynamic properties, and chemical species concentrations. It was determined that even at 300:1 the thermal efficiency of the cycle is still increasing for the multitude of hydrocarbon fuels investigated. At this point, for example, the efficiency with isooctane fuel was over 80%.

However, there are many engineering challenges involved in approaching the efficiency potential of ideal Otto cycle operation in real systems, specifically

1. The combustion process must be very rapid such that there is negligible piston motion through the process (this precludes diffusion controlled and flame propagation combustion);
2. The combustible mixture must be compressed to high levels before the burning process begins;
3. Heat loss from the combustion chamber must be minimized;
4. Fuel losses through the gas exchange process must be negligible.

Caris and Nelson (4) experimentally investigated the effect of high compression ratios on the thermal efficiency of homogeneous charge spark-ignition engines. By utilizing strong anti-knock additives, compression ratios of up to 24:1 were achieved. Their investigations revealed however, that the problems of finite burn duration (i.e. flame propagation), and thus non-constant-volume conditions limited the maximum efficiency point to a compression ratio of about 17:1.

Van Blarigan (5) experimentally investigated the potential of HCCI combustion operation to eliminate the problems of finite burn duration from the design considerations. By utilizing lean hydrogen-air mixtures (equivalence ratio = 0.3) under high compression (35:1) in a single-stroke combustion experiment, near constant-volume combustion conditions were achieved. It was concluded that under the proper conditions (based on the fuel, gas temperature, mixture strength, etc.) the compression ratio could be substantially increased and thus the thermal efficiency of the engine cycle improved.

Further experimental work at Sandia this year has continued to verify the potential of the HCCI combustion process for a number of fuels. The results have shown that the achievable compression ratio of the engine cycle is only a function of the autoignition characteristics of the fuel-air mixture. Very lean mixtures can be successfully ignited and burned, where this is impossible in conventional spark ignition hydrocarbon combustion systems. The NO_x emissions from the engine can be sufficiently reduced at these low equivalence ratios to the extent that only oxidation catalysts are required to control CO and HC emissions.

Further, the heat transfer losses from such low strength mixtures are reduced relative to stoichiometric operation since the maximum gas temperatures are lower. Finally, the free piston characteristics of the engine allow the compression ratio to be maximized for specific fuels, as the ignition delay time can be matched to the rate of compression at top dead center.

Figure 7 illustrates the differences in the free piston and crankshaft-driven piston dynamics near top dead center. It is obvious that much faster rates of compression are possible at the end of the stroke, and that the time spent at maximum compression is substantially decreased. The benefits of these characteristics are stated above.

Experimental Results.

Following are selected results of some of the experimental combustion studies completed this year. In these investigations a single-stroke rapid compression-expansion machine has been used to compression ignite homogeneous fuel-air mixtures of propane, natural gas, hydrogen, methanol, hexane, heptane and isooctane. Experimental details can be found in (5).

Figure 8 gives the efficiency and NO_x levels as a function of compression ratio for propane. Under these conditions, autoignition first occurs at a compression ratio of about 34:1. Figure 9 shows the hydrocarbon and carbon monoxide emissions for propane. Note that the hydrocarbon emissions are reported as parts per million of C_3H_y here and throughout this paper. The initial temperature and compression ratio have a large effect on the final emissions.

Results for the remaining fuels are presented in the same format. Figures 10, 11 and 12 are for natural gas. Note that for an initial temperature of 25C combustion does not occur until a compression ratio of 44:1 is - reached. This natural gas was a made-up blend of 93.12% methane, 3.2% ethane, 0.7% propane, 0.4% butane, 1.2% carbon dioxide and 1.37% nitrogen. Figures 13 and 14 show results for hydrogen, the fastest burning fuel. No hydrocarbon or carbon monoxide emissions are included.

The methanol test data is found in Figures 15, 16 and 17. The value of not taking too much data is seen here. Following methanol is pentane data in Figures 18, 19 and 20. The pentane tests were run late in our testing and show the consistency of later tests. Also shown in Figure 18 is an early energy release of a small fraction of the fuel, identified as the first ignition point.

The results for hexane shown in Figures 21, 22 and 23 are interesting in that the main reaction compression ratio is only 19:1 at an initial temperature of 25C. There also appears to be some reaction prior to the main combustion event as seen in Figure 21, where this reaction occurs at a compression ratio of about 16:1. This accounts for some of the poor showing in efficiency, but the hydrocarbon emissions are also large indicating not all of the fuel is reacting. The results for heptane shown in Figures 24, 25, 26 and 27 show a fuel that undergoes what appear to be three reaction points. Figures 24 and 25 show this in some detail. The final data set is for isooctane, shown in Figures 28, 29 and 30. A considerable amount of testing was done with this fuel due to the reduced performance at an initial temperature of 25C. The fuel reacted quickly, with no early reaction but did not seem to react completely. At 70C things were much better, although efficiency is still down from propane and emissions are higher.

Discussion

The data presented here were acquired to determine if rapid combustion at high compression ratio would produce high efficiency. Previous work in homogeneous charge spark ignition engines has consistently shown that indicated thermal efficiency does not increase with increasing compression ratio once 17:1 or so is reached, with some researchers reporting considerably lower rollover levels (6,7). To our knowledge, no investigators have measured performance contrary to this trend.

Our data certainly shows that it is possible to do better. From an ideal cycle with real gas effects perspective the potential improvement of a 30:1 compression ratio cycle with lean mixtures relative to a stoichiometric 12:1 compression ratio (about the limit of today's technology with a three way catalyst) cycle is 40% (9). Our experiments show a similar improvement relative to contemporary engine performance (7) with fuels such as propane, hydrogen and natural gas. Indicated efficiency of 56% has been measured with these fuels.

While we have presented all of the data, some of the tests are suspect. For example, we believe the efficiencies considerably above 56% are due to the combustion of seal lubricant from tests conducted immediately following new seal installation. We learned not to lubricate these parts. Similarly, some of the lower (and higher) results are from early tests before we were coating the pressure transducers to reduce thermal shock.

These tests, however, do not represent an engine. Intake/exhaust processes are not included, and the air/fuel mixture is completely quiescent before compression. Therefore, comparison to engine data must be done with great caution. Still, the data have shown the following trends:

1. High compression ratio at the time of combustion can be achieved. While initial temperature and fuel type have a strong effect on the compression ratio at reaction, ratios above 30:1 are possible at practical conditions. However, the data are not clear regarding efficiency improvement with increasing compression ratio. The relationship is confounded with the changes in starting temperature and fuels required to vary the reaction compression ratio.
2. The high combustion rate does approach constant volume combustion. Figure 13 shows a typical logarithmic P / V diagram for hydrogen combustion at top dead center at 33:1 compression ratio. The piston has, for all practical purposes, not moved during the combustion event, which figure 31 shows to take about 40 microseconds. In the free piston configuration high pressure-rise rates can be handled without difficulty since there are no load bearing linkages, as in crankshaft-driven engines. Additionally, operation at equivalence ratios less than 0.5 eliminates the need to consider piston erosion, or other physical damage (8).
3. Compressing the fuel/air mixture to higher levels after combustion occurs does not reduce thermal efficiency significantly. None of the data show a significant drop off in efficiency as the compression ratio is increased beyond that necessary to initiate combustion for constant initial conditions. However, the NO_x emissions do increase, a clear indication of increasing temperature or time at temperature. While it is possible that greater heat loss is compensated by more of the fuel reacting, this effect cannot be large (due to most of the fuel being reacted already) with the conclusion that heat loss does not increase significantly with higher compression ratio. A

contributor to this insensitivity is the coating of both the piston top and cylinder head with 0.25 mm of Silastic J to reduce the thermal conductivity of the surfaces. Figure 32 shows the uncoated cylinder head surface temperature during a combustion cycle. Coating of the head eliminates any measurable head surface temperature increase during this cycle. While all of the data presented were acquired under coated conditions, operation without the coating only reduced the efficiency by 5%.

4. NO_x emissions can be controlled by equivalence ratio to the desired level. C_3H_y and CO emissions are present in varying degree, but increasing the initial temperature generally reduces these emissions. We intend to investigate oxidation catalyst performance on these emissions. An interesting possibility for emissions control would be to utilize 50% internal EGR (i.e. leave 50% of the combustion products in the cylinder) and add a stoichiometric fresh charge of 50% of the cylinder volume. We performed such a test series and measured emissions after 4 cycles. NO_x was 130 PPM, CO was 1720 PPM and C_3H_y was 360 PPM. Efficiency was 50%. Such an operating strategy with a 3-way catalyst could be very attractive.
5. All fuels are not created equal. As the data show some fuels do not react completely or react in two steps. Generally, higher initial temperature and higher compression ratio reacted more of the fuel. All of the tests were conducted at the same compression/expansion rate (40 Hz oscillation rate). This is an interesting variable we intend to investigate in the future.

Other researchers are investigating homogeneous charge compression ignition in crankshaft engines (for example 10-14). Christensen et al (13, 14) evaluated isooctane, ethanol and natural gas in a 1.6 liter single cylinder displacement research engine at a fixed 21:1 compression ratio, at a speed of 1000 RPM. Their results are consistent with ours, with peak indicated efficiency (not including the inlet/exhaust strokes) of over 50% at similar equivalence ratio.

This raises the question as to whether the free piston geometry is important to this combustion concept. Certainly the lack of massive kinematic constraints is attractive for such high compression ratios, and the electronic control of compression ratio broadens the operating range. But the increased compression ratio possible with the free piston at the time of combustion may not provide much advantage. We intend to compare crankshaft and free piston performance under identical operating conditions to quantify the performance difference.

Finally, our work as well as Christensen's (13, 14) adds further credibility to the explanation of finite burn duration as the main cause for real cycle departure from ideal cycle performance as compression ratio is increased.

Summary

This section presents the results of an investigation which was conducted to determine the effect that homogeneous charge compression ignition of dilute fuel/air mixtures with a free piston would have on thermal efficiency and emissions. The investigation was conducted in a single stroke gas driven combustion experiment in which a premixed fuel/air charge was compressed to autoignition and expanded. Efficiency was calculated from measurements of pressure and piston displacement, and emissions were measured on the combustion gasses.

The results of this study have shown that indicated thermal efficiency significantly higher than is possible in spark ignition engines can be achieved. For example, the indicated thermal efficiency of hydrogen, propane or natural gas is 56%. The primary cause of this high conversion efficiency is nearly constant volume combustion at high compression ratio.

In addition this combustion approach controls NO_x formation by utilizing dilute mixtures, an approach not possible in spark ignition engines utilizing hydrocarbon fuels. Other regulated emissions must be dealt with by aftertreatment.

References

1. Robert H. Williams, "Fuel Decarbonization for Fuel Cell Applications and Sequestration of the Separated CO_2," PU/CEES Report No. 295, January 1996.

2. John B. Heywood, "Internal Combustion Engine Fundamentals," McGraw Hill, Inc. 1988, pp 712.

3. Edson, M. H., "The Influence of Compression Ratio and Dissociation on Ideal Otto Cycle Engine Thermal Efficiency," Digital Calculations of Engine Cycles, SAE Prog. in Technology, 7, 49-64. 1964.

4. Caris, D. F., and Nelson, N. N., "A New Look at High Compression Engines," SAE Transactions, **67**, 112-124, 1959.

5. Van Blarigan, P., "Advanced Hydrogen Fueled Internal Combustion Engines," Energy & Fuels, **12**, 72-77, 1998.

6. Overington, M. T. and Thring, R. H.: "Gasoline Engine Combustion - Turbulence and the Combustion Chamber," SAE Paper 810017, 1981.

7. Muranaka, Y. T. and Ishida, T.: "Factors Limiting the Improvement in Thermal Efficiency of S.I. Engine at Higher Compression Ratio," SAE Transactions, vol. 96, section 4, pp. 526 - 536, 1987.

8. Maly, R. R., et. al., "Theoretical and Experimental Investigation of Knock Induced Surface Destruction," SAE Transactions, **99**, Section 3, 99-137, 1990.

9. Edson, M. H., and Taylor, C. F.: "The Limits of Engine Performance - Comparison of Actual and Theoretical Cycles," *Digital Calculations of Engine Cycles, SAE Prog. in Technology*, vol. 7, pp.65-81, 1964.

10. Thring, R. H.: "Homogeneous-Charge Compression -Ignition Engines," SAE Paper 892068, 1989.

11. Najt, P. M. and Foster, D. E.: "Compression-Ignited Homogeneous Charge Combustion," SAE Paper 830264, 1983.

12. Karim, G. A. and Watson, H. C.: "Experimental and Computational Considerations of the Compression Ignition of Homogeneous Fuel-Oxidant Mixtures," SAE Paper 710133, 1971.

13. Christensen, M., Johansson, B., Amneus, P. and Mauss, F.: "Supercharged Homogeneous Charge Compression Ignition," SAE Paper 980787, 1998.

14. Christensen, M., Johansson, B. and Einewall, P.: "Homogeneous Charge Compression Ignition (HCCI) Using Isooctane, Ethanol and Natural Gas - A Comparison with Spark Ignition Operation," SAE Paper 972874, 1997.

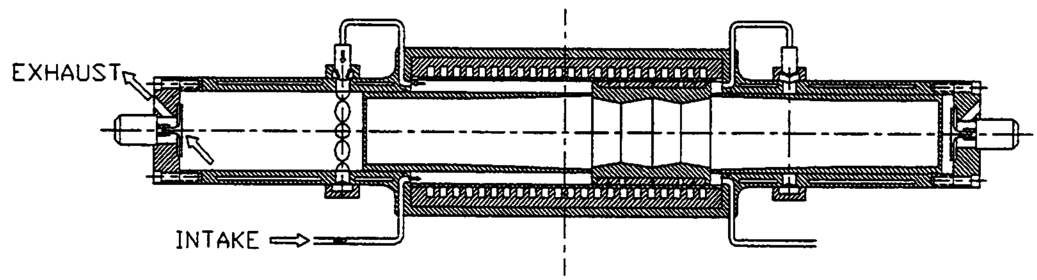


Figure 6 - Rapid Combustion Electrical Generator

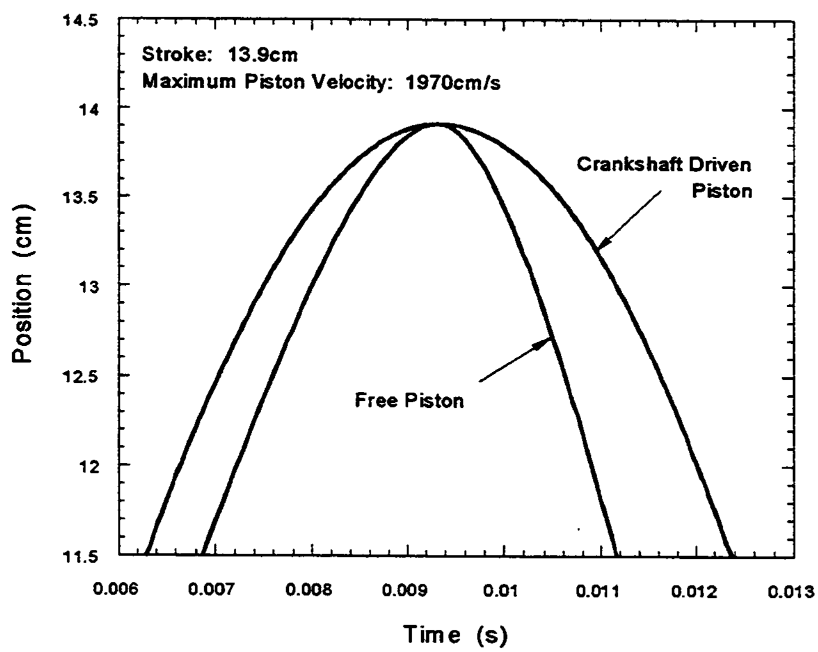
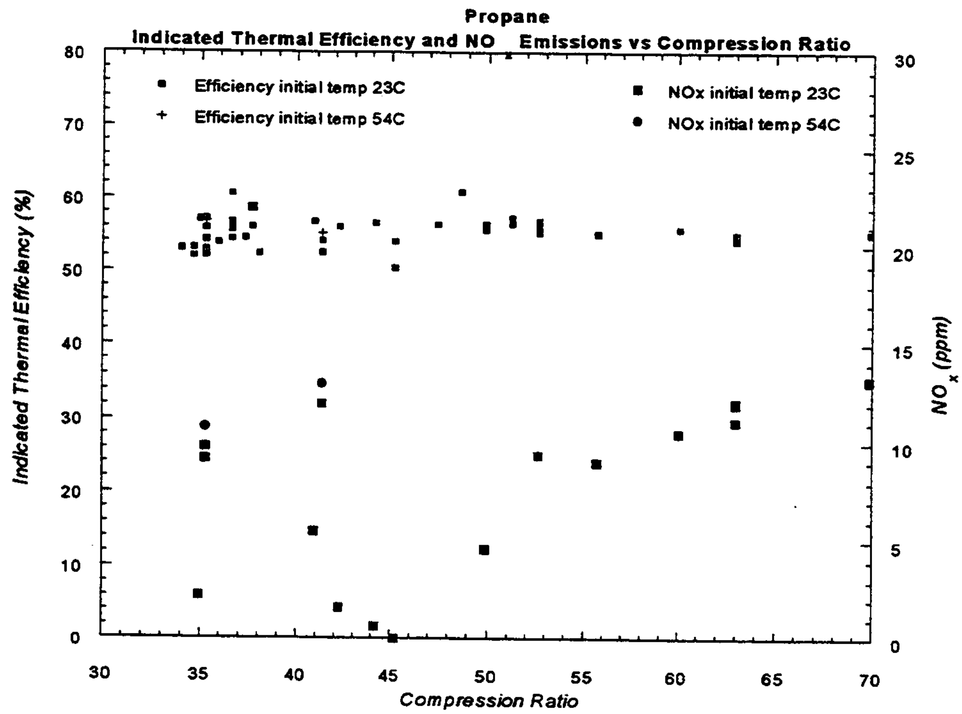


Figure 7 - Free Piston and Crankshaft Driven Piston Profiles



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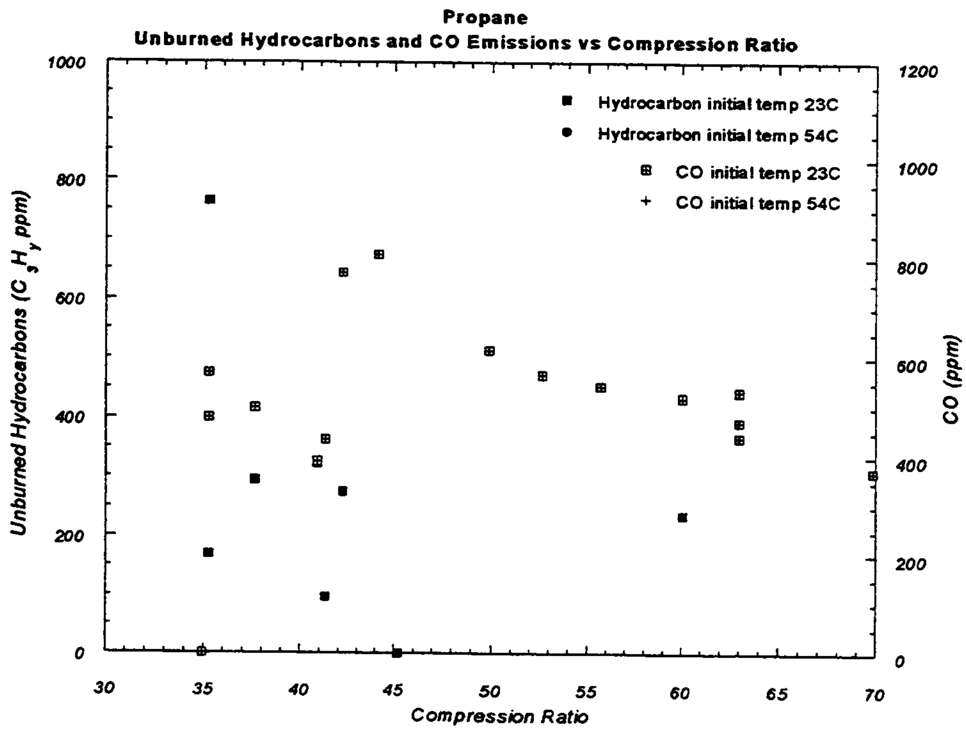

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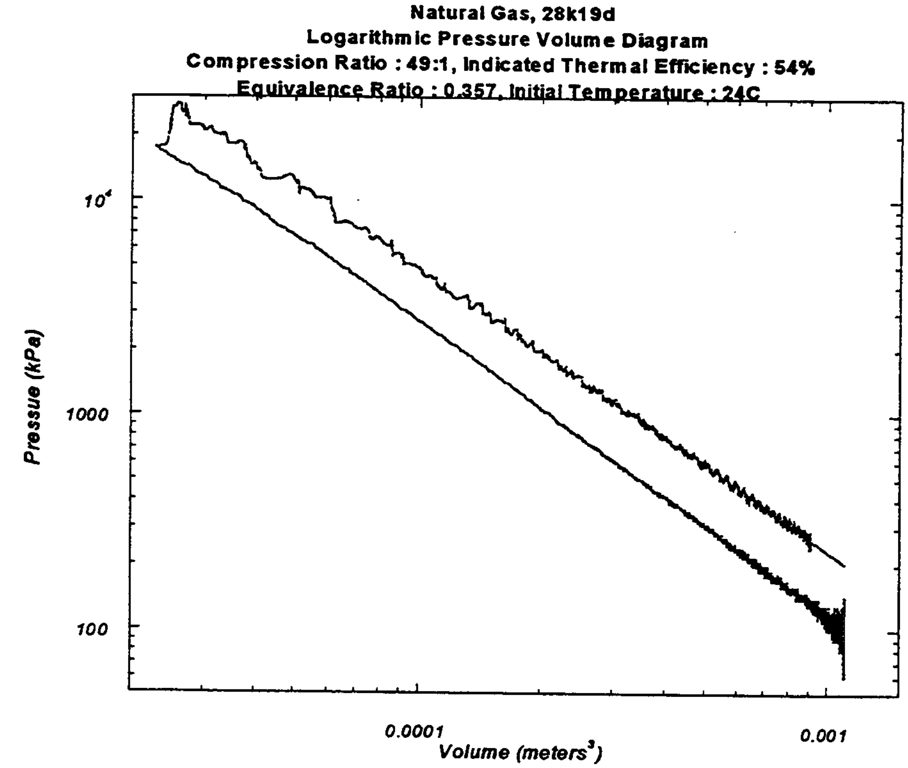

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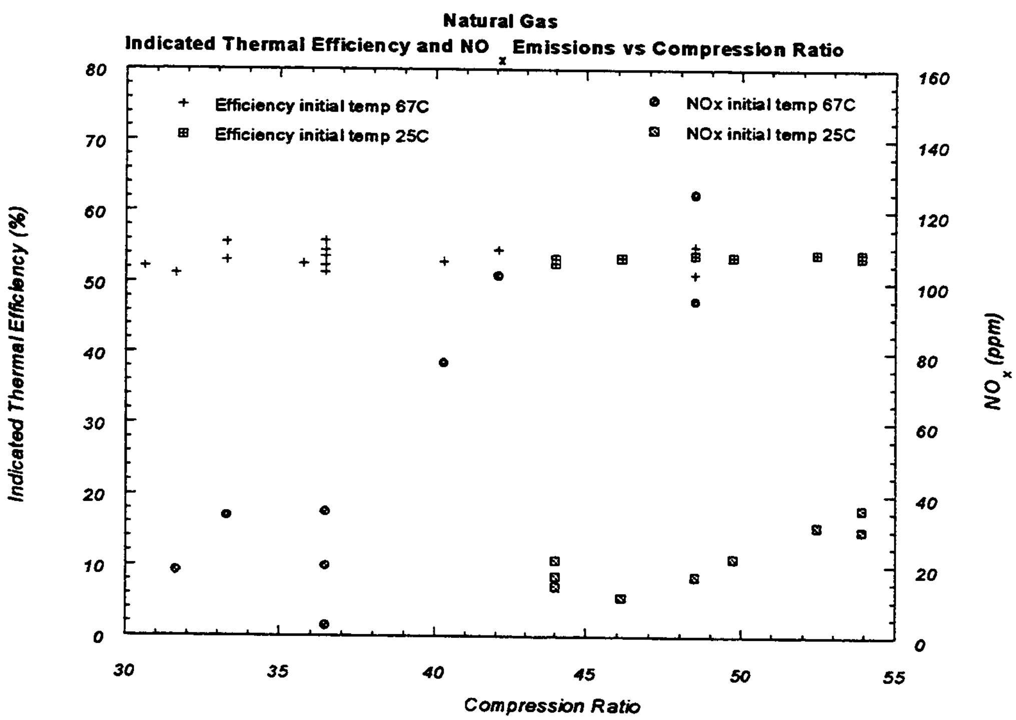

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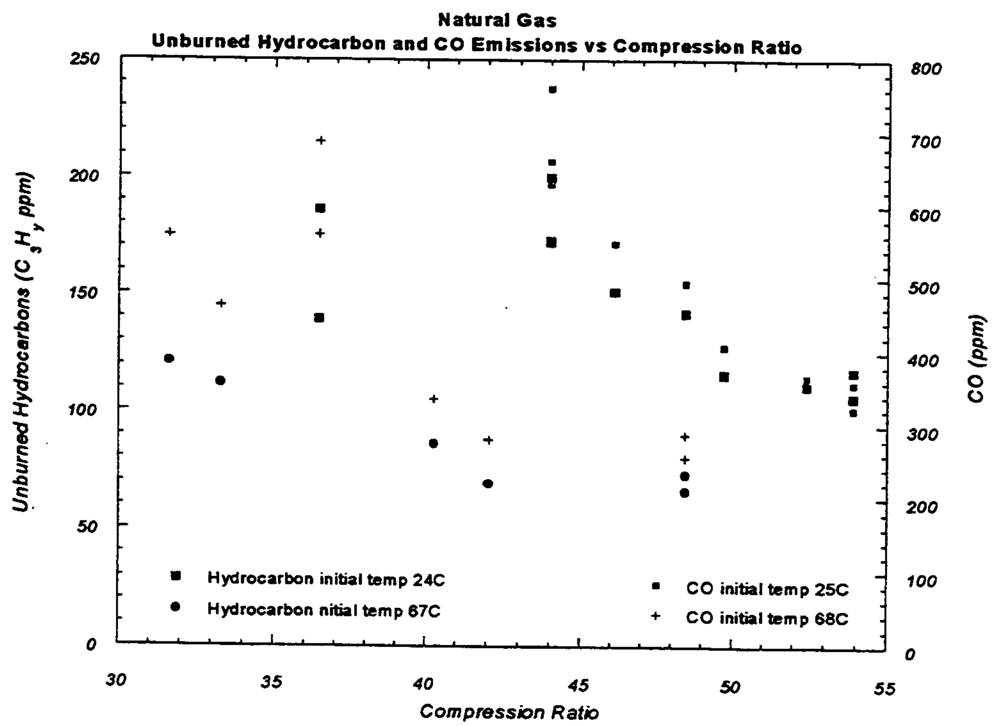

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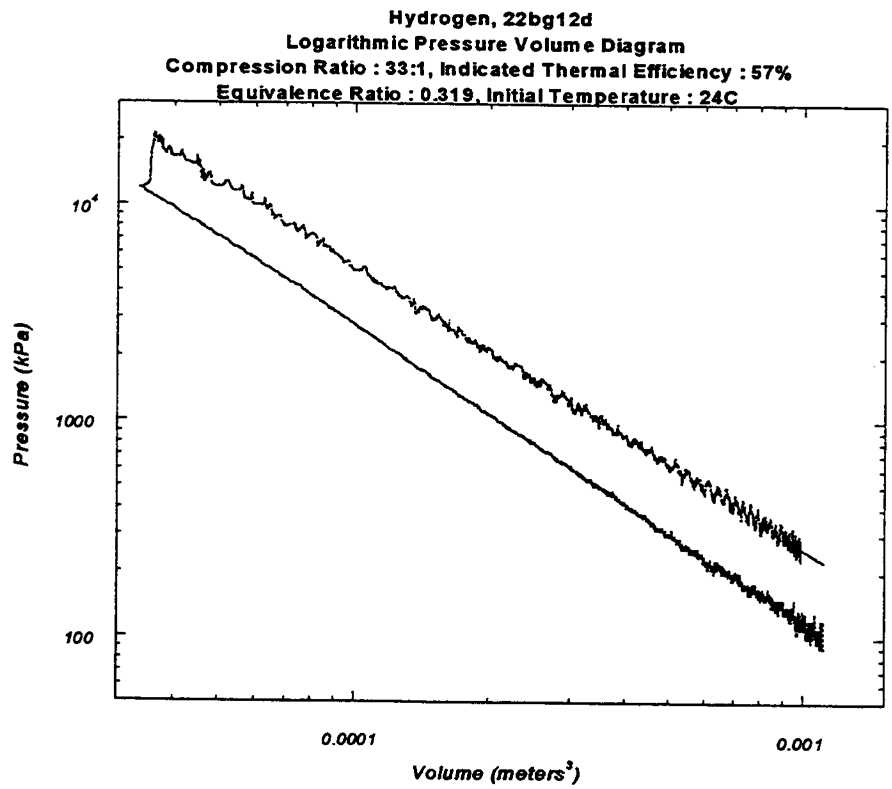

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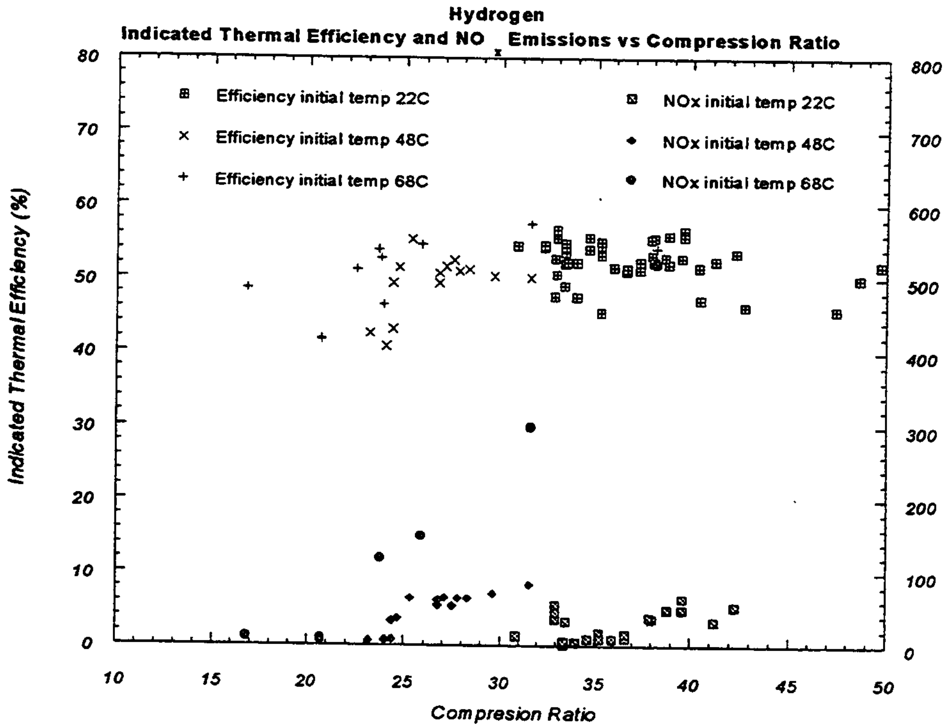

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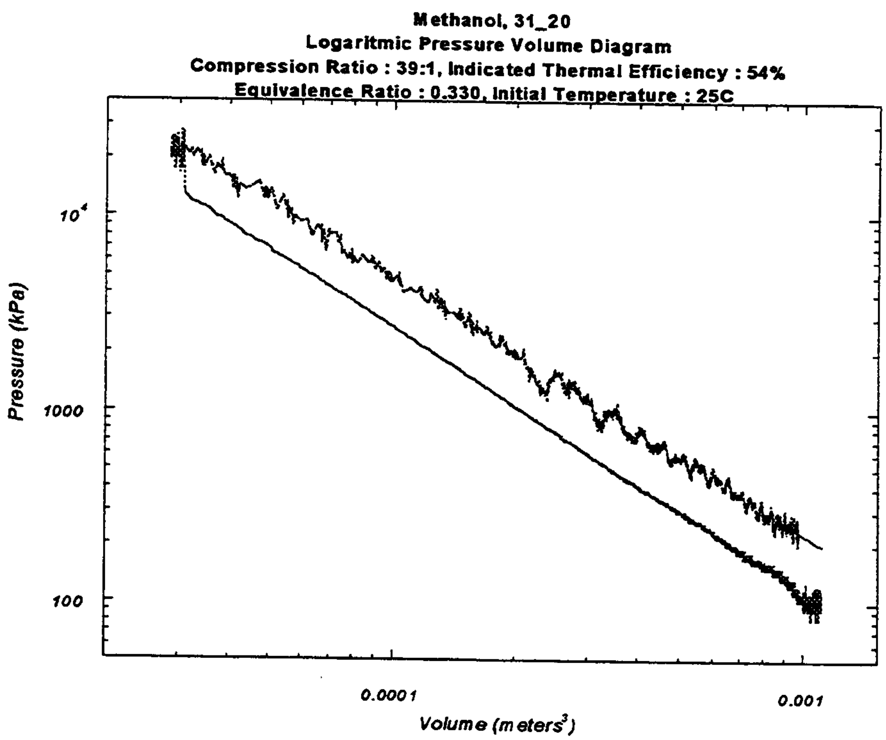

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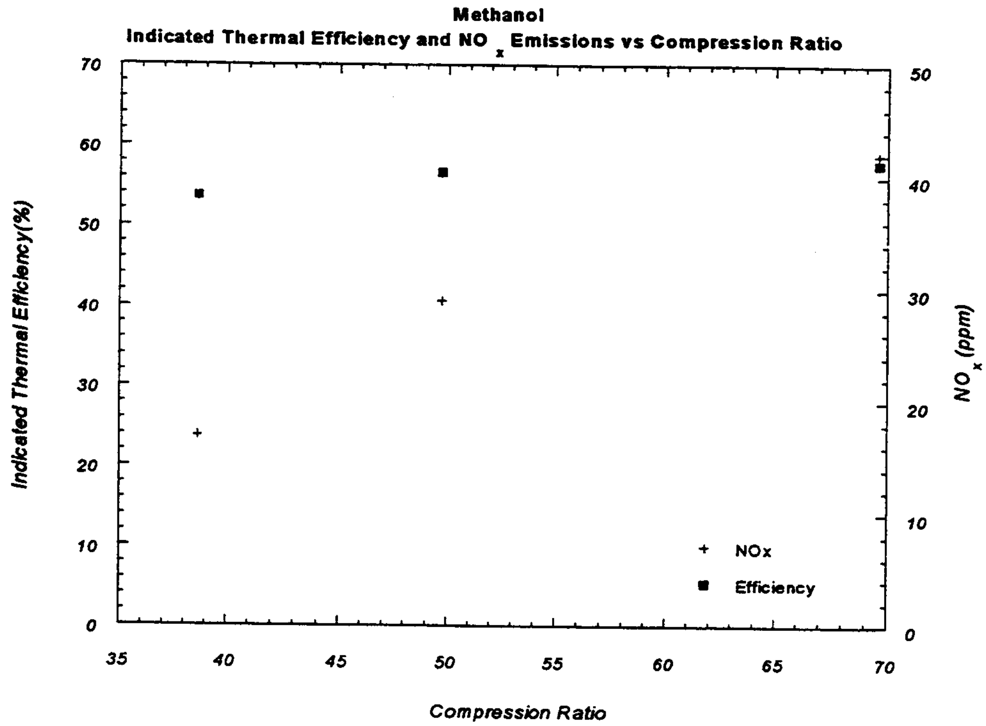

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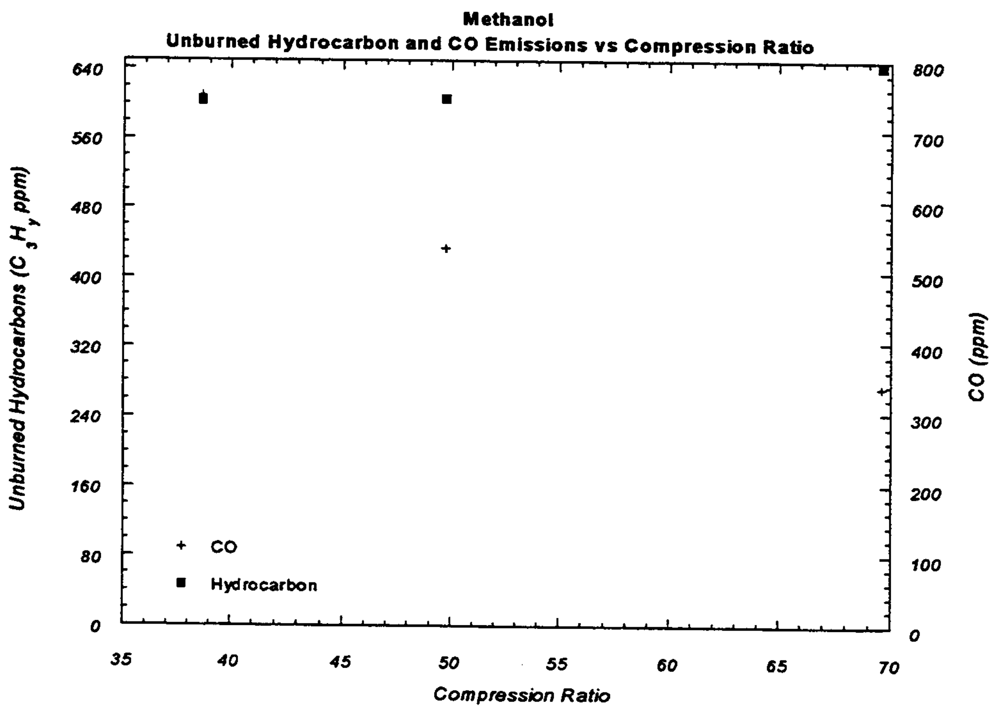

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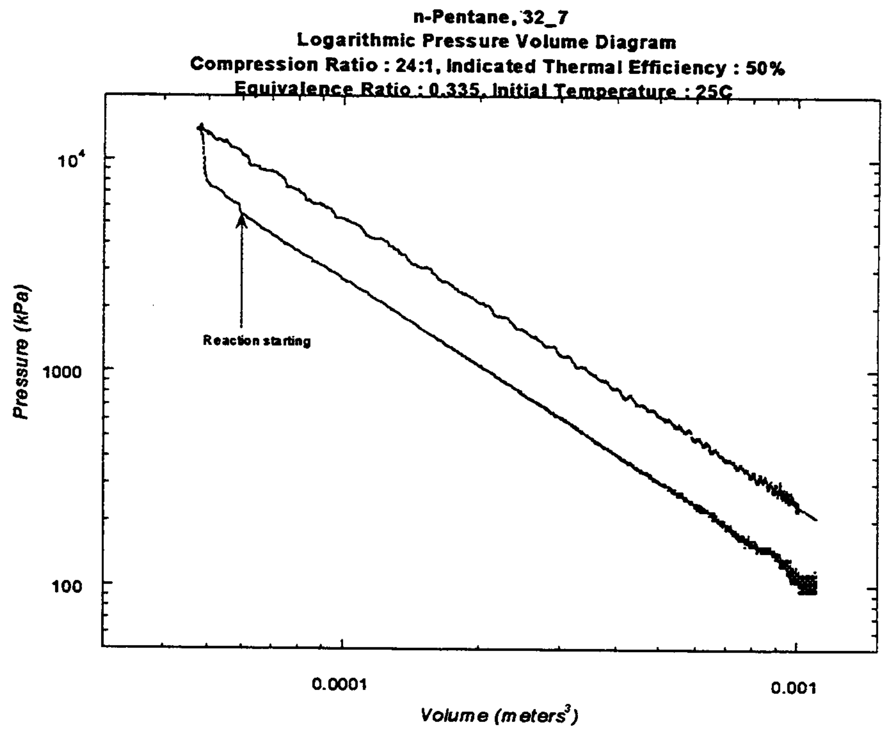

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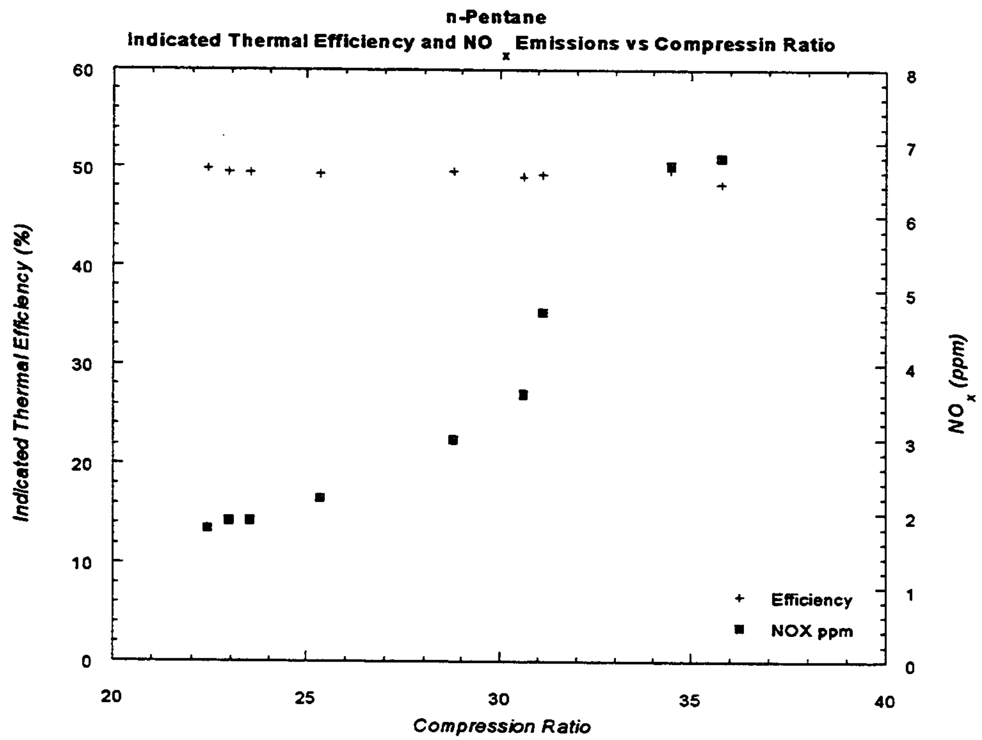

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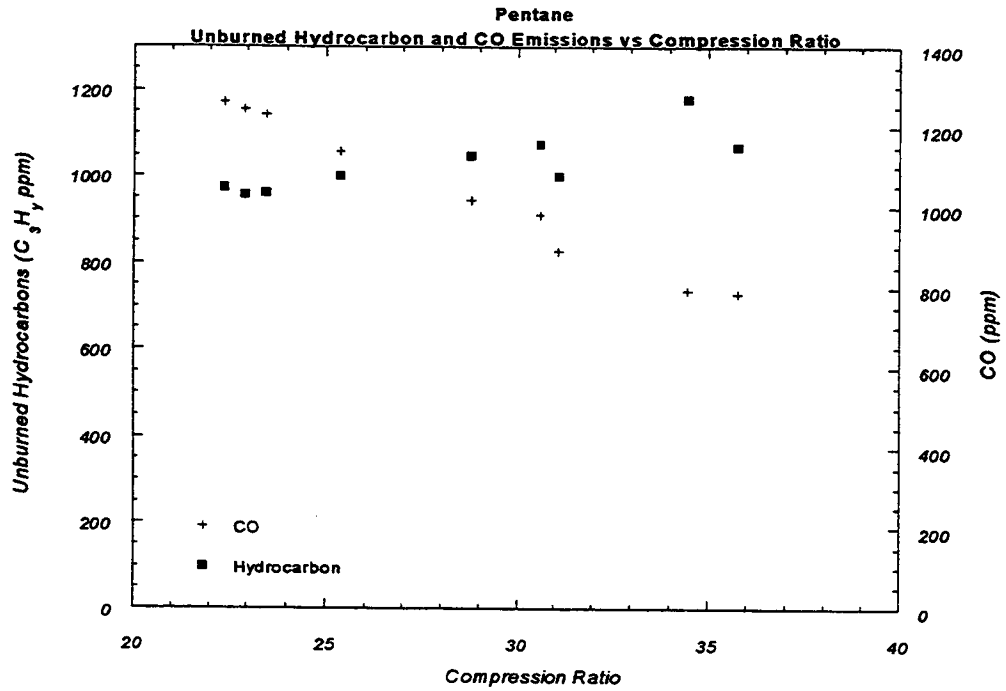

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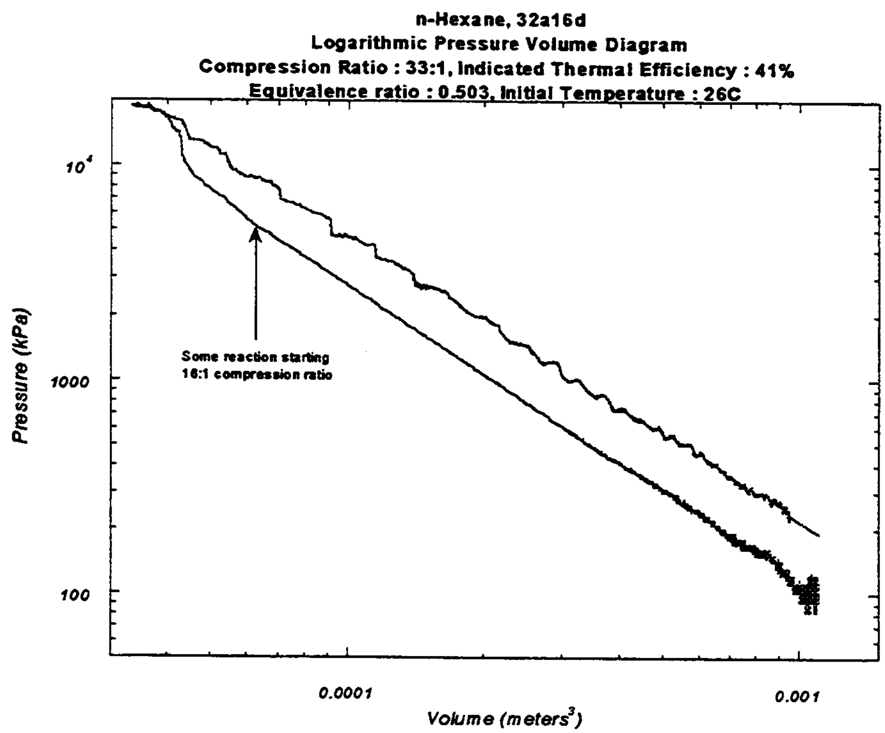

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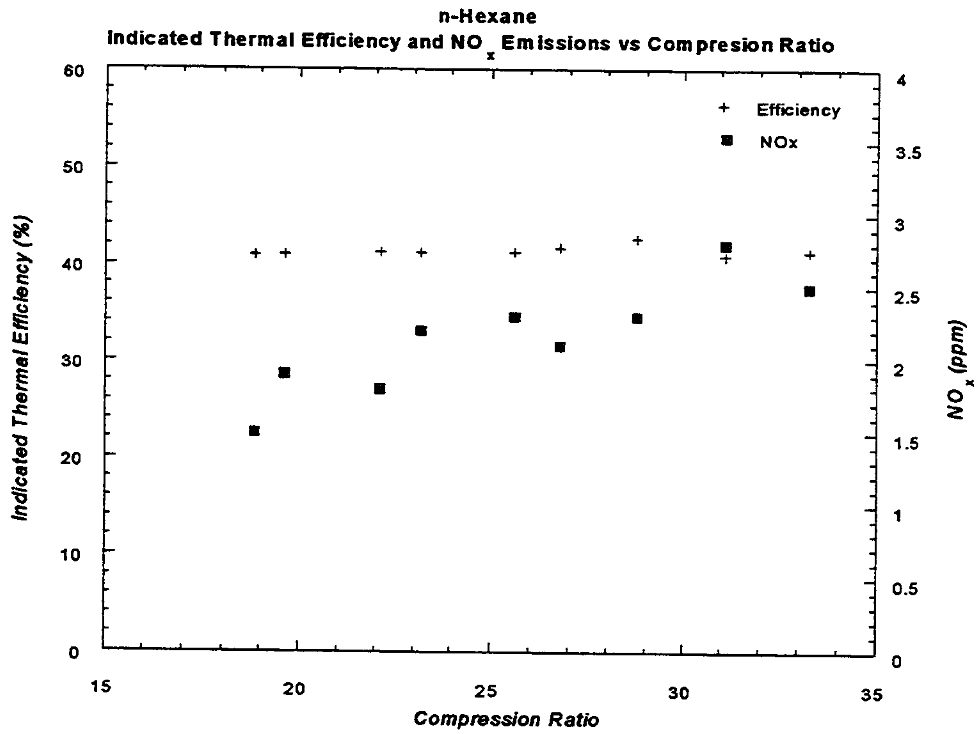

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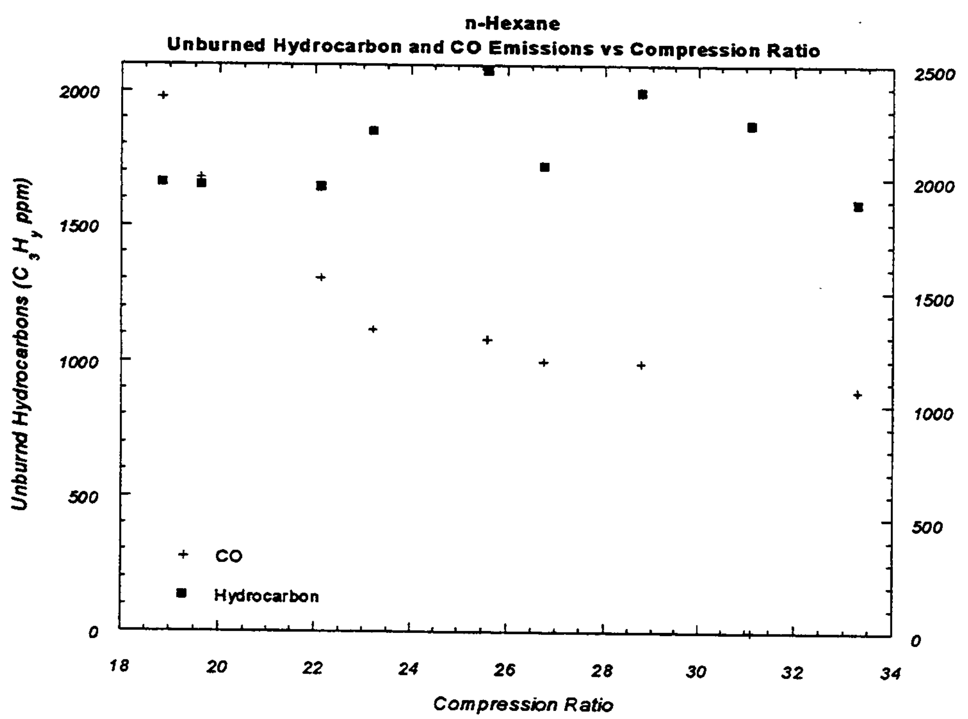

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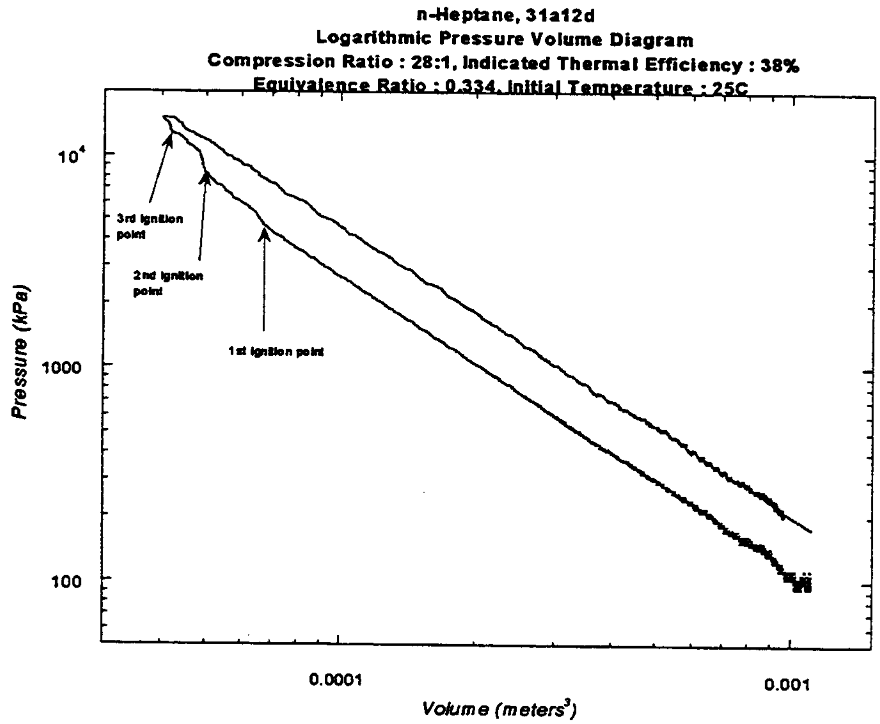

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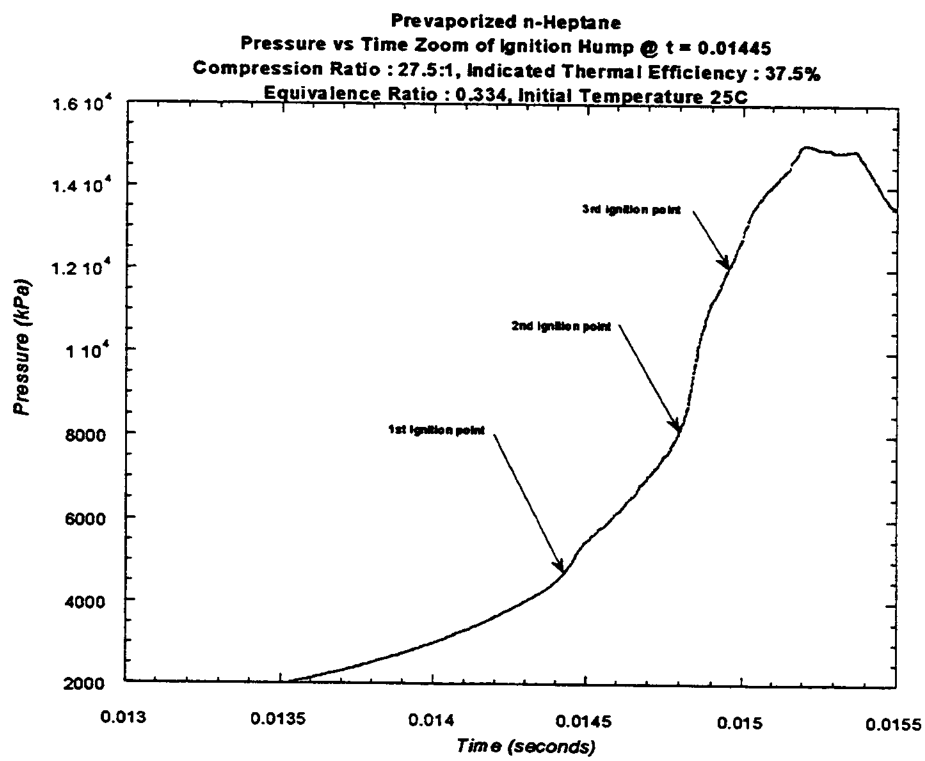

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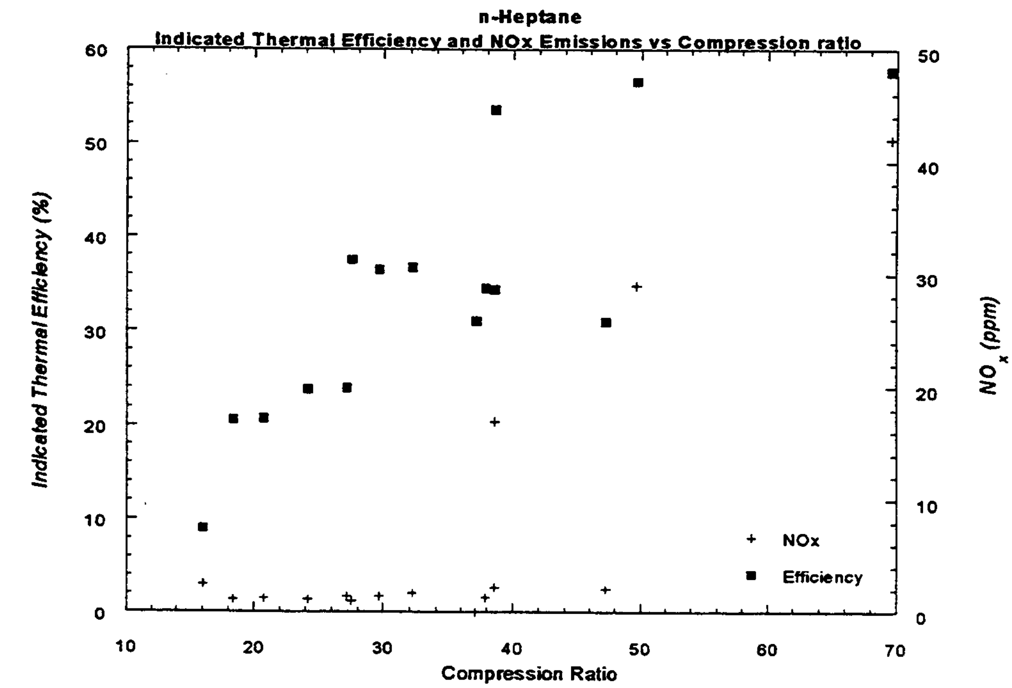

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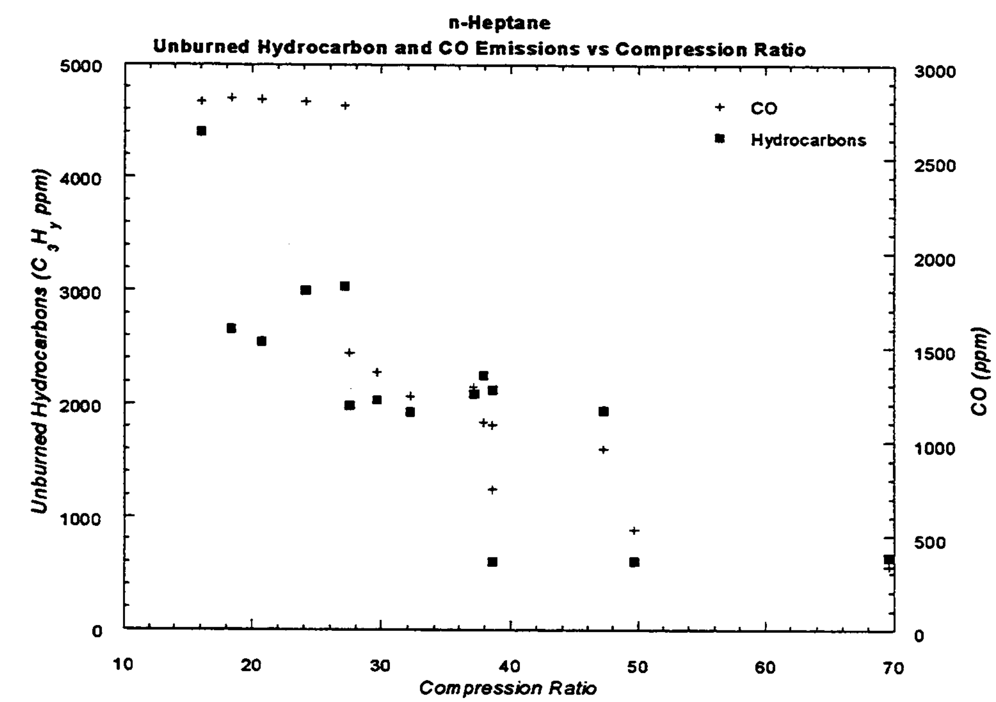

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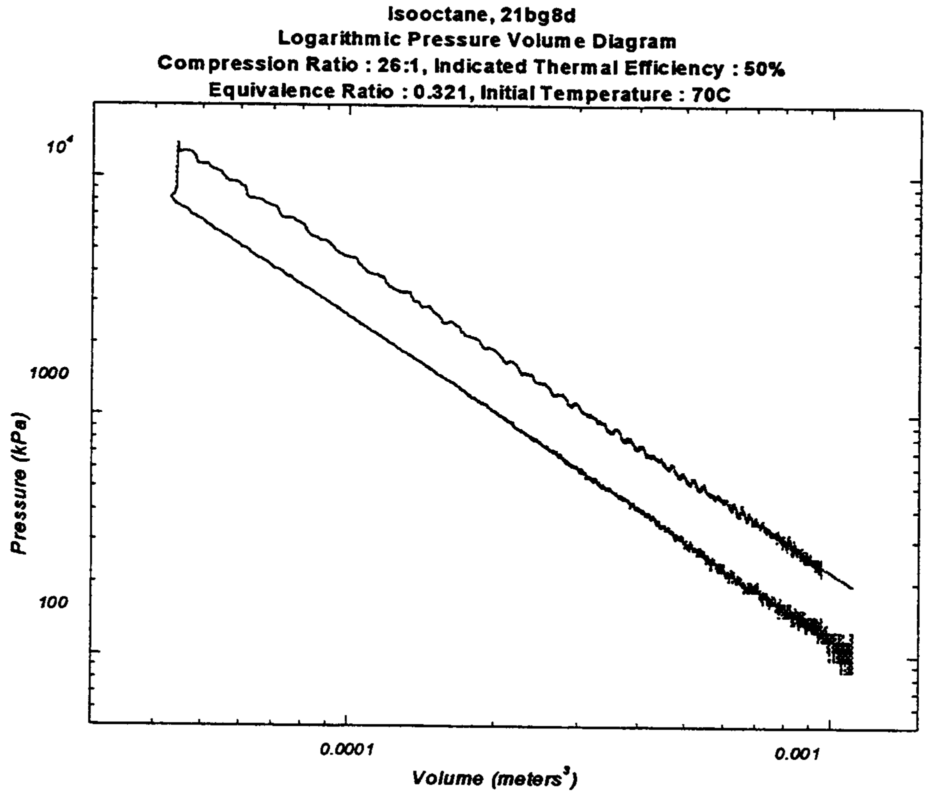

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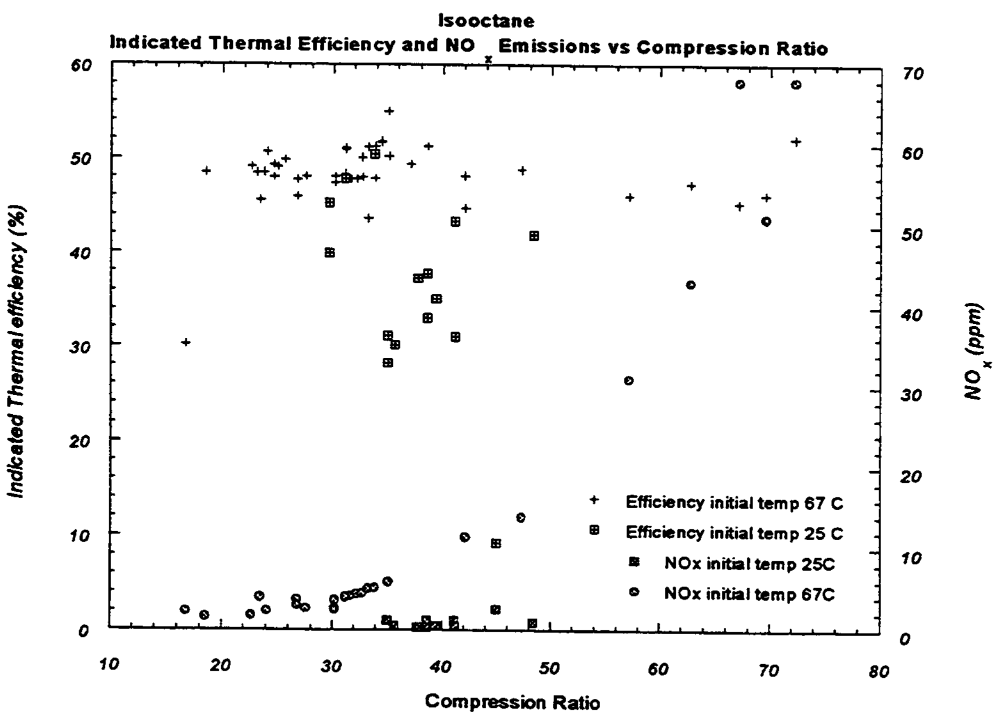

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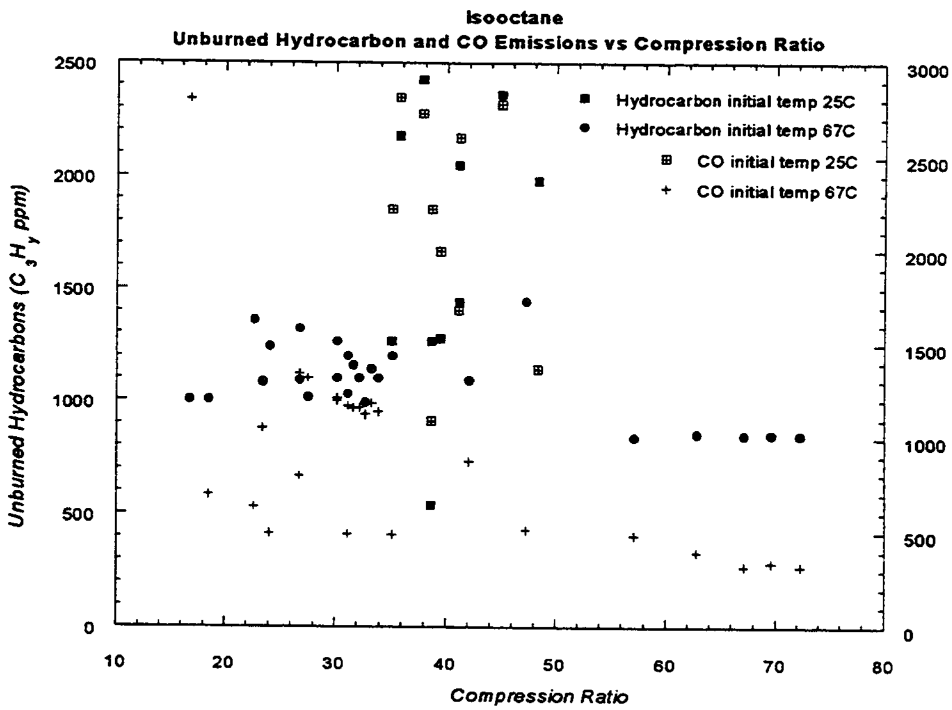

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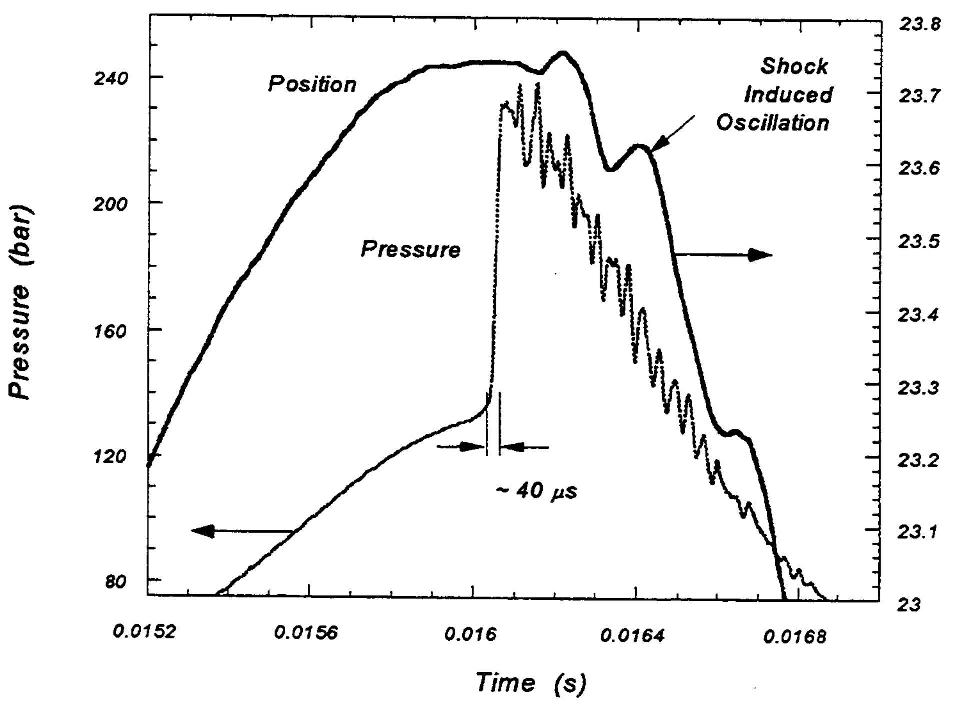

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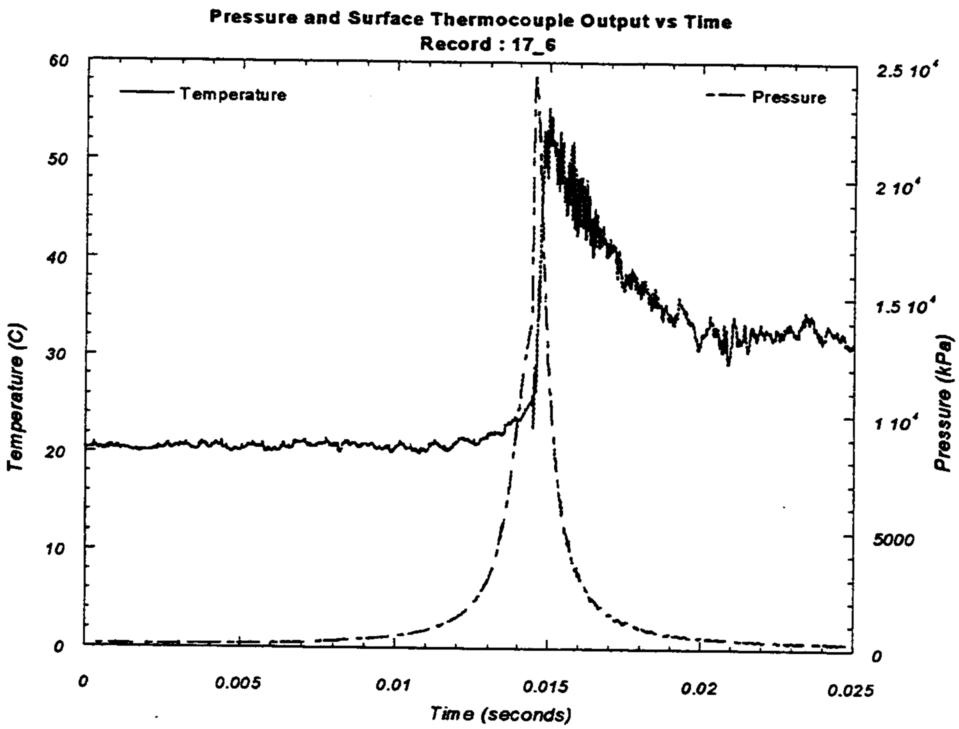
Pressure and Surface Thermocouple Output vs Time
Record : 17_6
Temperature
Pressure
Temperature (C)
Pressure (kPa)
Time (seconds)
0
10
20
30
40
50
60
5000
1 10⁴
1.5 10⁴
2 10⁴
2.5 10⁴
0.005
0.01
0.015
0.02
0.025

Figure 32